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(54) Variable displacement compressor

Kompressor mit veränderlicher Fördermenge

Compresseur à capacité variable

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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a variable displacement compressor for vehicle air-conditioning.

[0002] Fig. 8 shows a prior art variable displacement compressor according to a not published document of the inventor. Similar compressors are also known from for example US-A-5 332 365 or US-A-4 702 677. A drive shaft is rotatably supported in the housing 101, which encloses a crank chamber 102. A lip seal 104 is located between the housing 101 and the drive shaft 103 to prevent leakage of fluid from the housing 101.

[0003] An electromagnetic friction clutch 105 is located between the drive shaft 103 and the engine Eg, which serves as a power source. The clutch 105 includes a rotor 106 that is coupled to the engine Eg, an armature 107 that is fixed to the drive shaft 103, and an electromagnetic coil 108. When the coil 108 is excited, the armature 107 is attracted to and contacts the rotor 106. In this state, power of the engine Eg is transmitted to the drive shaft 103. When the coil 108 is de-excited, the armature 107 is separated from the rotor 106, which disconnects the power transmission from the engine Eg to the drive shaft 103.

[0004] A lug plate 109 is fixed to the drive shaft 103 in the crank chamber 102. A thrust bearing 122 is located between the lug plate 109 and the housing 101. A swash plate 110 is coupled to the lug plate 109 via a hinge mechanism 111. The swash plate 110 is supported by the drive shaft 103 such that the swash plate 110 slides axially and inclines with respect to the axis L of the drive shaft 103. The hinge mechanism 111 causes the swash plate 110 to integrally rotate with the drive shaft 103. When the swash plate 110 contacts the limit ring 112, the swash plate 110 is positioned at the minimum inclination position.

[0005] The housing 101 includes cylinder bores 113, a suction chamber 114, and a discharge chamber 115. A piston 116 is accommodated in each cylinder bore 113 and is coupled to the swash plate 110. A valve plate 117 partitions the cylinder bores 113 from a suction chamber 114 and a discharge chamber 115

[0006] When the drive shaft 103 rotates, the swash plate 110 reciprocates each piston 116. Accompanying this, refrigerant gas in the suction chamber 114 flows into each cylinder bore 113 through the corresponding suction port 117a and suction valve 117b, which are formed in the valve plate 117. Refrigerant gas in each cylinder bore 113 is compressed to reach a predetermined pressure and is discharged to the discharge chamber 115 through the corresponding discharge port 117c and discharge valve 117d, which are formed in the valve plate 117.

[0007] An axial spring 118 is located between the housing 101 and the drive shaft 103. The axial spring 118 urges the drive shaft 103 frontward (leftward in Fig. 8) along the axis L and limits axial chattering of the drive shaft 103. A thrust bearing 123 is located between the axial spring 118 and an end surface of the drive shaft 103. The thrust bearing 123 prevents transmission of rotation from the drive shaft 103 to the axial spring 118.

[0008] A bleed passage 119 connects the crank chamber 102 to the suction chamber 114. A pressurizing passage 120 connects the discharge chamber 115 to the crank chamber 102. A displacement control valve, which is an electromagnetic valve, adjusts the opening size of the pressurizing passage 120.

[0009] The control valve 121 adjusts the flow rate of refrigerant gas from the discharge chamber 115 to the crank chamber 102 by varying the opening size of the pressurizing passage 120. This varies the inclination of the swash pate 110, the stroke of each piston 116, and the displacement.

[0010] When the clutch 105 is disengaged, or when the engine Eg is stopped, the control valve 121 maximizes the opening size of the pressurizing passage 120. This increases the pressure in the crank chamber 102 and minimizes the inclination of the swash plate 110. As a result, the compressor stops when the inclination of the swash plate 110 is minimized, or when the displacement is minimized. Accordingly, since the displacement is minimized, the compressor is started with a minimal torque load. This reduces torque shock when the compressor is started.

[0011] When the cooling load on a refrigeration circuit that includes the compressor is great, for example, when the temperature in a vehicle passenger compartment is much higher than a target temperature set in advance, the control valve 121 closes the pressurizing passage 120 and maximizes the displacement of the compressor.

[0012] Suppose that when the compressor is operating at maximized displacement, it is stopped by disengagement of the clutch 105 or by shutting off the engine Eg. In this case, the control valve 121 quickly maximizes the opening size of the closed pressurizing passage 120 to minimize the displacement. Also, when the vehicle is suddenly accelerated while the compressor is operating at maximum displacement, the control valve 121 quickly maximizes the opening size of the pressurizing passage 120 to minimize the displacement and to reduce the load applied to the engine Eg. Accordingly, refrigerant gas in the discharge chamber 115 is quickly supplied to the crank chamber 102. Though some refrigerant gas flows to the suction chamber 114 through the bleed passage 119, the pressure in the crank chamber 102 quickly increases.

[0013] Therefore, the swash plate 110, when at a minimum displacement position (as shown by the broken line in Fig. 8) is pressed against a limit ring 112. Also, the swash plate 110 pulls the lug plate 109 in a rearward direction (rightward in Fig. 8) through the hinge mechanism 111. As a result, the drive shaft 103 moves axially rearward against the force of the axial spring 118.

[0014] When the drive shaft 103 moves rearward, the axial position of the drive shaft 103 with respect to a lip seal 104, which is held in the housing 101, changes. Generally, a predetermined contact area of the drive shaft 103 contacts the lip seal 104. Foreign particles such as sludge exist on the peripheral surface of the drive shaft 103 that is outside the predetermined contact area. Therefore, when the axial position of the drive shaft 103 with respect to the lip seal 104 changes, the sludge will be located between the lip seal 104 and the drive shaft 102. This lowers the sealing performance of the lip seal 104 and may cause leakage of refrigerant gas from the crank chamber 102.

[0015] When the operation of the compressor is stopped by the disengagement of the clutch 105 and the drive shaft 103 moves rearward, the armature 107, which is fixed to the drive shaft 103, moves toward the rotor 106. The clearance between the rotor 106 and the armature 107 when the clutch 105 is disengaged is set to a small value, for example, 0.5mm. Accordingly, when the drive shaft 103 moves rearward, the clearance between the rotor 106 and the armature 107 is eliminated, which causes the armature 107 to contact the rotating rotor 106. This may cause noise and vibration or may transmit power from the engine Eg to the drive shaft 103 regardless of the disengagement of the clutch 105.

[0016] When the drive shaft 103 moves rearward, each piston 116, which is coupled to the drive shaft through the lug plate 109 and the swash plate 110, also moves rearward. This moves the top dead center position of each piston 116 toward the valve plate 117 which may permit the pistons 116 to collide with the valve plate 117. Since the control valve 121 maximizes the opening size of the pressurizing passage 120 during sudden accelerations of the vehicle while the compressor is operating, the rearward movement of the drive shaft 103 accompanying the control may cause the pistons 116 to repeatedly collide with the valve plate 117. This generates noise and vibration.

[0017] To prevent the rearward movement of the drive shaft 103, the force of the axial spring 118 can be increased. However, increasing the force of the axial spring 118 lowers the durability of the thrust bearing 123, which is located between the axial spring 118 and the drive shaft 103, lowers the durability of the thrust bearing 122, which is located between the housing 101 and the lug plate 109, and increases the load placed on the engine by the compressor.

SUMMARY OF THE INVENTION

[0018] An objective of the present invention is to provide a variable displacement compressor that can prevents the pressure in a crank chamber from excessively increasing.

[0019] To achieve the above objective, the present invention provides a variable displacement compressor comprising the features of claim 1.

[0020] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0021] The features of the present invention are set forth with particularity in the appended claims. The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a cross sectional view showing a variable displacement compressor according to a first embodiment of the present invention;

Fig. 2 is a cross sectional view showing the displacement control valve of the compressor of Fig. 1;

Fig. 3 is a partial enlarged cross-sectional view showing the electromagnetic friction clutch of the compressor of Fig. 1;

Fig. 4 is a partial enlarged view showing the release valve of the compressor of Fig. 1;

Fig. 5 is a cross sectional view showing a variable displacement compressor according to a second embodiment;

Fig. 6 is a partial enlarged cross-sectional view showing a release valve in a third embodiment;

Fig. 7 is a partial enlarged cross-sectional view showing a release valve in a fourth embodiment; and

Fig. 8 is a cross sectional view of a prior art variable displacement compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0022] A single head type variable displacement compressor for vehicle air-conditioners according to a first embodiment of the present invention will now be described with reference to Figs. 1-4.

[0023] As shown in Fig. 1, a front housing member 11 and a rear housing member 13 are coupled to a cylinder block 12. A valve plate 14 is located between the cylinder block 12 and the rear housing member 13. The front housing member 11, the cylinder block 12, and the rear housing member form a compressor housing.

[0024] As shown in Figs. 1 and 2, the valve plate 14 includes a main plate 14a, a first sub-plate 14b, a sec-

ond sub-plate 14c, and a retainer plate 14d. The main plate 14a is located between the first sub-plate 14b and the second sub-plate 14c. The retainer plate 14d is located between the second sub-plate 14c and the rear housing member 13.

[0025] A crank chamber 15 is defined between the front housing member 11 and the cylinder block 12. A drive shaft 16 passes through the crank chamber 15 and is rotatably supported by the front housing member 11 and the cylinder block 12.

[0026] The drive shaft 16 is supported in the front housing member 11 through the radial bearing 17. A central bore 12a is formed substantially in the center of the cylinder block 12. The rear end of the drive shaft 16 is located in the central bore 12a and is supported in the cylinder block 12 through the radial bearing 18. A spring seat 21, which is a snap ring, is fixed to the inner surface of the central bore 12a. The thrust bearing 19 and the axial spring 20 are located in the central bore 12a between the rear end surface of the drive shaft 16 and the spring seat 21. The axial spring 20, which is a coil spring, urges the drive shaft frontward (leftward in Fig. 1) through the thrust bearing 19. The axial spring 20 is an urging member. The thrust bearing 19 prevents transmission of rotation from the drive shaft 16 to the axial spring 20.

[0027] The front end of the drive shaft 16 projects from the front housing member 11. A lip seal 22, which is a shaft sealing assembly, is located between the drive shaft 16 and the front housing member 11 to prevent leakage of refrigerant gas along the surface of the drive shaft 16. The lip seal 22 includes a lip ring 22a, which is pressed against the surface of the drive shaft 16.

[0028] An electromagnetic friction clutch 23 is located between an engine Eg, which serves as an external power source, and the drive shaft 16. The clutch 23 selectively transmits power from the engine Eg to the drive shaft 16. The clutch 23 includes a rotor 24, a hub 27, an armature 28, and an electromagnetic coil 29. The rotor 24 is rotatably supported by the front end of the front housing member 11 through an angular bearing 25. A belt 26 is received by the rotor 24 to transmit power from the engine Eg to the rotor 24. The hub 27, which has elasticity, is fixed to the front end of the drive shaft 16 and supports the armature 28. The armature 28 is arranged to face the rotor 24. The electromagnetic coil 29 is supported by the front wall of the front hosing member 11 to face the armature 28 across the rotor 24.

[0029] When the coil 29 is excited while the engine Eg is running, an attraction force based on electromagnetic force is generated between the armature 28 and the rotor 24. Accordingly, the armature 28 contacts the rotor 24, which engages the clutch 23. When the clutch 23 is engaged, power from the engine Eg is transmitted to the drive shaft 16 through the belt 26 and the clutch 23 (See Fig. 1). When the coil 29 is de-excited in this state, the armature 28 is separated from the rotor 24 by the elasticity of the hub 27, which disengages the clutch 23. When the clutch 23 is engaged, transmission of power from the engine Eg to the drive shaft 16 is disconnected (See Fig. 3).

[0030] As shown in Fig. 1, a lug plate 30 is fixed to the drive shaft 16 in the crank chamber 15. A thrust bearing 67 is located between the lug plate 30 and the inner wall of the front housing member 11. A swash plate 31, which serves as a drive plate, is supported on the drive shaft 16 to slide axially and to incline with respect to the drive shaft 16. A hinge mechanism 32 is located between the lug plate 30 and the swash plate 31. The swash plate 31 is coupled to the lug plate 30 through the hinge mechanism 32. The hinge mechanism 32 integrally rotates the swash plate 31 with the lug plate 30. The hinge mechanism 32 also guides the swash plate 31 to slide along and incline with respect to the drive shaft 16. As the swash plate 31 moves toward the cylinder block 12, the inclination of the swash plate 31 decreases. As the swash plate 31 moves toward the lug plate 30, the inclination of the swash plate 31 increases.

[0031] A limit ring 34 is attached to the drive shaft 16 between the swash plate 31 and the cylinder block 12. As shown by the broken line in Fig. 1, the inclination of the swash plate 31 is minimized when the swash plate 31 abuts against the limit ring 34. On the other hand, as shown by solid lines in Fig. 1, the inclination of the swash plate 31 is maximized when the swash plate 31 abuts against the lug plate 30.

[0032] Cylinder bores 33 are formed in the cylinder block 12. The cylinder bores 33 are arranged at equal annular intervals about the axis L of the drive shaft 16. A single head piston 35 is accommodated in each cylinder bore 33. Each piston 35 is coupled to the swash plate 31 through a pair of shoes 36. The swash plate 31 converts rotation of the drive shaft 16 into reciprocation of the pistons 35.

[0033] A suction chamber 37 , which is a suction pressure zone, is defined in the substantial center of the rear housing member 13. A discharge chamber 38, which is a discharge pressure zone, is formed in the rear housing member 13 and surrounds the suction chamber 37 . The main plate 14a of the valve plate 14 includes suction ports 39 and discharge ports 40, which correspond to each cylinder bore 33. The first sub-plate 14b includes suction valves 41, which correspond to suction ports 39. The second sub-plate 14c includes discharge valves 42, which correspond to the discharge ports 40. The retainer plate 14d includes retainers 43, which correspond to the discharge valves 42. Each retainer 43 determines the maximum opening size of the corresponding discharge valve 42.

[0034] When each piston 35 moves from the top dead center position to the bottom dead center position, refrigerant gas in the suction chamber 37 flows into the corresponding cylinder bore 33 through the corresponding suction port 39 and suction valve 41. When each piston 35 moves from the bottom dead center position to the top dead center position, refrigerant gas in the

corresponding cylinder bore 33 is compressed to a predetermined pressure and is discharged to the discharge chamber 38 through the corresponding discharge port 40 and discharge valve 42.

[0035] A pressurizing passage 44 connects the discharge chamber 38 to the crank chamber 15. A bleed passage 45, which is a pressure release passage, connects the crank chamber 15 to the suction chamber 37 . The bleed passage 45 functions as a control passage that connects the crank chamber 15 to a selected chamber in the compressor, which is the suction chamber 37 in this embodiment. A displacement control valve 46 is located in the pressurizing passage 44. The control valve 46 adjusts the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15 by varying the opening size of the pressurizing passage 44. The bleed passage 45 and the control valve 46 form a pressure control mechanism. The pressure in the crank chamber 15 is varied in accordance with the relation between the flow rate of refrigerant from the discharge chamber 38 to the crank chamber 15 and that from the crank chamber 15 to the suction chamber 37 through the bleed passage 45. Accordingly, the difference between the pressure in the crank chamber 15 and the pressure in the cylinder bores 33 is varied, which varies the inclination of the swash plate 31. This varies the stroke of each piston 35 and the displacement.

[0036] The control valve 46 will now be described.

[0037] As shown in Fig. 2, the control valve 46 includes a valve housing 65 and a solenoid 66, which are coupled together. A valve chamber 51 is defined between the valve housing 65 and the solenoid 66. The valve chamber 51 accommodates a valve body 52. A valve hole 53 opens in the valve chamber 51 and faces the valve body 52. An opener spring 54 is accommodated in the valve chamber 51 and urges the valve body 52 to open the valve hole 53. The valve chamber 51 and the valve hole 53 form part of the pressurizing passage 44.

[0038] A pressure sensitive chamber 55 is formed in the valve housing 65. The pressure sensitive chamber 55 is connected to the suction chamber 37 through a pressure detection passage 47. A bellows 56, which is a pressure sensitive member, is accommodated in the pressure sensitive chamber 55. A spring 57 is located in the bellows 56. The spring 57 determines the initial length of the bellows 56. The bellows 56 is coupled to and operates the valve body 52 through a pressure sensitive rod 58, which is integrally formed with the valve body 52.

[0039] A plunger chamber 59 is defined in the solenoid 66. A fixed iron core 60 is fitted in the upper opening of the plunger chamber 59. A movable iron core 61 is accommodated in the plunger chamber 59. A follower spring 62 is located in the plunger chamber 59 and urges the movable core 61 toward the fixed core 60. A solenoid rod 63 is integrally formed at the lower end of the valve body 52. The distal end of the solenoid rod 63 continuously abuts against the movable core 61 by the forces of the opener spring 54 and the follower spring 62. In other words, the valve body 52 moves integrally with the movable core 61 through the solenoid rod 63. The fixed core 60 and the movable core 61 are surrounded by a cylindrical electromagnetic coil 64.

[0040] As shown in Fig. 1, the suction chamber 37 is connected to the discharge chamber 38 through an external refrigerant circuit 71. The external refrigerant circuit 71 includes a condenser 72, an expansion valve 73, an evaporator 74. The external refrigerant circuit 71 and the variable displacement compressor constitute a refrigeration circuit.

[0041] A controller C is connected to an air-conditioner switch 80, which is a main switch of the vehicle air-conditioner, a temperature adjuster 82 for setting a target temperature in a passenger compartment, and a gas pedal sensor 83. The controller C is, for example, a computer, which is located on current supply lines between a power source S (a vehicle battery) and the clutch 23 and between the power source S and the control valve 46. The controller C supplies electric current from the power source S to the electromagnetic coils 29, 64. The controller C controls current supply to each coil 29, 64 based on information including the ON/Off state of the air-conditioner switch 80, a temperature detected by the temperature sensor 81, a target temperature set by the temperature adjuster 82, and the gas pedal depression degree detected by the gas pedal sensor 83.

[0042] When the engine Eg is stopped (when the ignition switch is positioned at the accessory off position), most of the current supply to the electric equipment of the vehicle is stopped. Accordingly, the supply of current from the power source S to each coil 29, 64 is stopped. That is, when the operation of the engine Eg is stopped, the current supply lines between the power source S and each coil 29, 64 are disconnected upstream of the controller C.

[0043] Operation of the control valve 46 will now be described.

[0044] The controller C supplies a predetermined electric current to the coil 29 of the clutch 23 when the air-conditioner switch 80 is turned on during the operation of the engine Eg, and the temperature detected by the temperature sensor 81 is higher than the target temperature set by the temperature adjuster 82. This engages the clutch 23 and starts the compressor.

[0045] The bellows 56 of the control valve 46 is displaced in accordance with the pressure in the suction chamber 37 , which is connected to the pressure sensitive chamber 55. The displacement of the bellows 56 is transmitted to the valve body 52 through the pressure sensitive rod 58. On the other hand, the controller C determines the electric current value supplied to the coil 64 of the control valve 46 based on the temperature detected by the temperature sensor 81 and the target temperature set by the temperature adjuster 82. When an electric current is supplied to the coil 64, an electromag-

netic attraction force in accordance with the value of the current is generated between the fixed core 60 and the movable core 61. The attraction force is transmitted to the valve body 52 through the solenoid rod 63. Accordingly, the valve body 52 is urged to reduce the opening size of the valve hole 53 against the force of the opener spring 54.

[0046] In this way, the opening size of the valve hole 53 by the valve body 52 is determined by the equilibrium of the force applied from the bellows 56 to the valve body 52, the attraction force between the fixed core 60 and the movable core 61, and the force of each spring 54, 62.

[0047] As the cooling load on the refrigeration circuit increases, for example, as the temperature detected by the temperature sensor 81 becomes higher than the target temperature set by the temperature adjuster 82, the controller C instructs the control valve 46 to increase the current supply to the coil 64. This increases the attraction force between the fixed core 60 and the movable core 61 and increases the force that urges the valve body 52 toward the closed position of the valve hole 53. In this case, the bellows 56 operates the valve body 53 targeting a relatively low suction pressure. In other words, as the current supply increases, the control valve 46 adjusts the displacement of the compressor to maintain a relatively low suction pressure (corresponding to a target suction pressure).

[0048] As the opening size of the valve hole 53 is reduced by the valve body 52, the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15 through the pressurizing passage 44 is reduced. On the other hand, refrigerant gas in the crank chamber 15 continuously flows to the suction chamber 37 through the bleed passage 45. This gradually decreases the pressure in the crank chamber 15. Accordingly, the difference between the pressure in the crank chamber 15 and the pressure in the cylinder bores 33 is decreased, which increases the inclination of the swash plate 31 and the displacement of the compressor.

[0049] As the cooling load on the refrigeration circuit decreases, for example, as the difference between the temperature detected by the temperature sensor 81 and the target temperature set by the temperature adjuster 82 decreases, the controller C reduces the current supply to the coil 64. This weakens the attraction force between the fixed core 60 and the movable core 61 and reduces the force that urges the valve body 52 toward the closed position of the valve hole 53. In this case, the bellows 56 operates the valve body 52 targeting a relatively high suction pressure. In other words, as the current supply decreases, the control valve 46 adjusts the displacement of the compressor to maintain a relatively high suction pressure (corresponding to a target suction pressure).

[0050] As the opening size of the valve hole 53 increases, the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15 is increased, which gradually increases the pressure in the crank chamber 15. This increases the difference between the pressure in the crank chamber 15 and the pressure in the cylinder bores 12a and reduces the inclination of the swash plate 31 and the displacement of the compressor.

[0051] A structural characteristic of the present embodiment will now be described.

[0052] As shown in Fig. 1, a pressure release passage 90 is independent from the bleed passage 45 and connects the crank chamber 15 to the suction chamber 37. The release passage 90 functions as a control passage, which connects the crank chamber 15 to a selected chamber, which is the suction chamber 37 in this embodiment. As shown in Figs. 1 and 4, a release valve 95, which is an electromagnetic valve in this embodiment, is located in the release passage 90. The release valve 95 includes a solenoid 95a, which is controlled by the controller C, and a valve body 95b, which varies the opening size of the release passage 90. When the solenoid 95a is excited, the valve body 95b closes the release passage 90 (See Fig. 1). When the solenoid 95a is de-excited, the valve body 95b opens the release passage 90 (See Fig. 4).

[0053] When the air-conditioner switch 80 is turned off during the operation of the compressor, the controller C stops the current supply to the coil 29 and disengages the clutch 23 and simultaneously stops the current supply to the coil 64 of the control valve 46. Further, the controller C stops the current supply to the solenoid 95a of the release valve 95.

[0054] When the gas pedal depression degree, which is detected by the gas pedal sensor 83, is greater than a predetermined value during the operation of the compressor, the controller C judges that the vehicle is being quickly accelerated and stops the current supply to the coil 64 of the control valve 46 and to the solenoid 95a of the release valve 95 for a predetermined period.

[0055] When the engine Eg is stopped during the operation of the compressor, the current supply lines between the power source S and each coil 29, 64 and between the power source S and the solenoid 95a are disconnected upstream of the controller C. Accordingly, the current supply to the coil 29 is stopped and the clutch 23 is disengaged, which stops the current supply to the coil 64 and the solenoid 95a.

[0056] When the clutch 23 is disengaged or the engine Eg is stopped, the current supply to the coil 64 of the control valve 46 is stopped. Then, the attraction force between the fixed core 60 and the movable core 61 disappears, and the control valve 46 fully opens the pressurizing passage 44. This increases the pressure in the crank chamber 15 and minimizes the inclination of the swash plate 31. As a result, the compressor is stopped when the inclination of the swash plate 31 is minimized, or when the displacement is minimized. Accordingly, since the compressor is started from the minimum displacement state, which produces a minimum torque load, the torque shock of starting the compressor

is limited.

[0057] When the gas pedal depression degree detected by the gas pedal sensor 83 is greater than a predetermined value, the current supply to the coil 64 is stopped. This causes the control valve 46 to fully open the pressurizing passage 44. As a result, the inclination of the swash plate 31 is minimized and the compressor is operated at the minimum displacement with relatively low torque load. Therefore, the load on the engine Eg is reduced and the vehicle is smoothly accelerated.

[0058] When the current supply to the coil 64 is stopped while the compressor is operated at maximum displacement, the control valve 46 quickly maximizes the opening size of the closed pressurizing passage 44. This permits relatively high-pressure refrigerant gas in the discharge chamber 38 to flow quickly to the crank chamber 15. Since the amount of refrigerant gas that flows from the crank chamber 15 to the suction chamber 37 through the bleed passage 45 and the through hole 91a of the release valve 91 is limited, the pressure in the crank chamber 15 is quickly increased.

[0059] However, when the pressure in the crank chamber 15 increases to an excessive degree by the discontinuation of the current supply to the coil 64, the current supply to the solenoid 95a is simultaneously stopped, which causes the release valve 95 to open the release passage 90 as shown in Fig. 4. Therefore, a relatively large amount of gas flows from the crank chamber 15 to the suction chamber 37 through the release passage 90. As a result, an excessive increase of the pressure in the crank chamber 15 is limited, which prevents the swash plate from being pressed against the limit ring 34 by an excessive force when at the minimum inclination position. Also, the swash plate 31 does not strongly pull the lug plate 30 rearward (rightward in Fig. 1) through the hinge mechanism 32. As a result, the drive shaft does not move axially against the force of the axial spring 20.

[0060] When the vehicle is quickly accelerated while the compressor is operating at maximum displacement, the load on the engine Eg can be reduced by disengaging the clutch 23. However, shock is produced when engaging or disengaging the clutch 23, which lowers the vehicle performance. However in this embodiment, the clutch 23 is not disengaged when the vehicle is quickly accelerated, which improves the vehicle performance.

[0061] The present embodiment has the following advantages.

[0062] Excessive increases of the pressure in the crank chamber 15 are prevented by opening the electromagnetic release valve 95 in the release passage 90. As a result, the drive shaft 16 is prevented from moving axially against the force of the axial spring 20.

[0063] The drive shaft 16 does not move with respect to the lip seal 22. That is, the position of the drive shaft 16 with respect to the lip ring 22a of the lip seal 22 does not change. Therefore, sludge does not get in the space between the lip ring 22a and the drive shaft 16. This extends the life of the lip seal 22 and prevents leakage of gas from the crank chamber 15.

[0064] The armature 28 of the clutch 23 moves with respect to the rotor 24 in the direction of axis L and contacts or separates from the rotor 24. In the present embodiment, since the axially rearward movement of the drive shaft 16 is prevented, a desirable clearance is ensured between the rotor 24 and the armature 28 when the clutch 23 is disengaged. Accordingly, power transmission between the rotor 24 and the armature 28 is disrupted without fail while the electromagnetic coil 29 of the clutch 23 is de-excited. This prevents noise, vibration, and heat that are caused by contact between the rotor 24 and the armature 28.

[0065] Each piston 35 is connected to the drive shaft 16 through the lug plate 30, the hinge mechanism 32, the swash plate 31 and the shoes 36. The axially rearward movement of the drive shaft 16 is prevented, which prevents the pistons 35 from moving toward the valve plate 14. As a result, the pistons 35 are prevented from colliding with the valve plate 14 at the top dead center position. Therefore, noise and vibration caused by the collision between the piston 35 and the valve plate 14 are suppressed.

[0066] The opening size of the pressurizing passage 44 is varied by controller C based on the information including the passenger compartment temperature, the target temperature, and the gas pedal depression degree. Compared to a compressor having a control valve that operates in accordance with only suction pressure, a sudden change of displacement from the maximum to the minimum can occur in the compressor including the control valve 46, that is, the pressure in the crank chamber 15 can be quickly increased. Therefore, the release valve 95 of the compressor of Fig. 1 effectively prevents sudden increases of the pressure in the crank chamber 15.

[0067] Compared to a pressure difference valve that opens or closes the release passage 90 according to a difference of pressure between the crank chamber 15 and the suction chamber 37, the release valve 95, which is an electromagnetic valve operated by external instructions, responsively opens the release passage 90 without fail. Accordingly, the release valve 95 limits the pressure in the crank chamber 15.

[0068] When the current supply to the coil 64 of the control valve 46 is stopped, the current supply to the solenoid 95a is simultaneously stopped and the valve body 95 opens the release passage 90. In other words, the pressure in the crank chamber 15 when the pressurizing passage is fully opened is limited by opening the release passage 90. This is an advantage of the electromagnetic release valve 95, which cannot be achieved by the pressure difference valve.

[0069] The control valve 46 varies the displacement of the compressor by changing the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15 by changing the opening size of the pres-

surizing passage 44. The compressor of Fig. 1 can more quickly increase the pressure in the crank chamber 15 than a compressor that only adjusts the flow of refrigerant from the crank chamber 15 to the suction chamber 37 to vary the displacement. Accordingly, when the compressor is stopped, the displacement is quickly minimized. When the compressor is restarted right after the previous stop, the compressor is started at the minimum displacement without fail. The release valve 95 is especially effective for the compressor of Fig. 1, which tends to excessively increase the pressure in the crank chamber 15.

[0070] For example, the structure of the control valve 46 may be changed such that the attraction force between the fixed core 60 and the movable core 61 operates the valve body 52 to increase the opening size of the valve hole 53. In this case, the current supply from the power source S to the coil 64 must be maximized to minimize the displacement especially when the engine Eg is stopped. In other words, it is necessary to maintain the current supply line between the power source S and the coil 64. This requires a drastic change from the existing electrical system.

[0071] In contrast, the control valve 46 of the present embodiment only stops the current supply from the power source S to the coil 64 to minimize the displacement when the engine Eg is stopped. Accordingly, it does not matter that the current supply line between the power source S and the coil 64 is disconnected when the engine Eg is stopped. Therefore, the displacement is minimized without changing the structure of existing vehicle electric systems.

[0072] The illustrated embodiments can be varied as follows.

[0073] As shown in Fig. 5, the valve body 95b may not completely close the release passage 90 when the solenoid 95a is excited. This permits restricted gas flow through the space between the release passage 90 and the valve body 95b when the difference between the pressure in the crank chamber 15 and the pressure in the suction chamber 37 is smaller than predetermined value. Therefore, the release passage 90 releases gas from the crank chamber 15 with restriction and prevents an excessive increase of the pressure in the crank chamber 15. Accordingly, the bleed passage 45 is not required.

[0074] As shown in Fig. 6, a through hole 95c that is smaller than the cross-sectional area of the release passage 90 may be formed in the valve body 95b of the release valve 95. When the difference between the pressure in the crank chamber 15 and the pressure in the suction chamber 37 is smaller than predetermined value, or when the solenoid 95a is excited, the through hole 95c releases gas from the crank chamber 15 in a restricted manner. Therefore, the release passage 90 releases gas from the crank chamber 15 and prevents an excessive increase of the pressure in the crank chamber 15. Therefore, the bleed passage 45 is not required.

[0075] As shown in Fig. 7, instead of the release passage 90, a pressure limiting passage 100, which limits the pressure in the crank chamber 15, may be provided between the discharge chamber 38 and the crank chamber 15. The release valve 95 is located in the pressure limiting passage 100. The pressure limiting passage 100 is independent from the pressurizing passage 44. When the pressure in the crank chamber 15 increases excessively, the release valve 95 decreases the opening size of or completely closes the pressure limiting passage 100, which limits the supply of refrigerant gas to the crank chamber 15.

[0076] As shown in Fig. 1, the release valve 95 may open the release passage 90 only when the current supply to the coil 64 is stopped while the compressor is operated at the maximum displacement. In other words, when the current supply to the coil 64 is stopped while the compressor is operating at the maximum displacement, the release valve 95 is not opened.

[0077] In any of the embodiments shown in Figs. 1-4, when the gas pedal depression increases, the controller C judges that the vehicle is being quickly accelerated. Instead, the controller C may judge that the vehicle is being quickly accelerated when the engine speed of the engine Eg is greater than a predetermined value.

[0078] The present invention may be applied to a compressor that varies the displacement by adjusting the flow of refrigerant gas from the crank chamber 15 to the suction chamber 37 by the control valve 46. In this case, the control valve 46 is located in a passage that connects the crank chamber 15 to the suction passage 37.

[0079] The present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope of the appended claims.

Claims

1. A variable displacement compressor comprising:

   a housing including a cylinder bore (33), a crank chamber (15), a suction chamber (37), and a discharge chamber (38);
   a piston (35) accommodated in the cylinder bore (33);
   a drive shaft (16) rotatably supported in the housing;
   a drive plate (31) coupled to the piston (35) for converting rotation of the drive shaft (16) to reciprocation of the piston (35), the drive plate (31) being tiltably supported on the drive shaft (16), wherein the drive plate (31) moves between a maximum inclination position and a minimum inclination position in accordance

with the pressure in the crank chamber (15), wherein the inclination of the drive plate (31) determines the piston (35) stroke and the displacement of the compressor;
a pressure control mechanism (44, 46) for controlling the pressure in the crank chamber (15) to change the inclination of the drive plate (31), wherein the pressure control mechanism (44, 46) comprises a control valve (46), which has a solenoid (66); and
a control passage (90, 100) for connecting the crank chamber (15) to a selected chamber in the compressor, the compressor **being characterized by:**

a pressure adjusting valve (95) located in the control passage (90, 100), wherein the pressure adjusting valve (95) has a solenoid (95a), wherein the pressure adjusting valve (95) regulates gas flow in the control passage (90, 100); and
a controller for controlling the current supplied to the solenoid (95a) of the pressure adjusting valve (95) to limit the pressure in the crank chamber (15) to prevent the pressure in the crank chamber (15) from becoming undesirably high.

2. The compressor according to claim 1, **characterized in that** the compressor includes an urging member (20) that urges the drive shaft (16) in an axial direction, which restricts axial movement of the drive shaft (16), wherein the pressure in the crank chamber (15) causes the drive plate (31) to apply an axial force to the drive shaft (16) when the drive plate (31) is located at the minimum inclination position, wherein the controller instructs the pressure adjusting valve (95) to limit the pressure in the crank chamber (15) such that the axial force cannot move the drive shaft (16) against the force of the urging member.

3. The compressor according to claim 1, **characterized in that** the pressure control mechanism (44, 46) includes a pressurizing passage for connecting the discharge chamber (38) to the crank chamber (15), wherein the control valve (46) is located in the pressuring passage, wherein the control valve (46) controls a flow of gas from the discharge chamber (38) to the crank chamber (15) through the pressurizing passage, wherein the control valve substantially fully opens the pressurizing passage to move the drive plate (31) to the minimum inclination position based on commands from the controller.

4. The compressor according to claim 1, **characterized in that** the selected chamber is the suction chamber (37), wherein the control passage (90) allows gas to flow from the crank chamber (15) to the suction chamber (37), wherein the controller opens the pressure adjusting valve (95) to increase gas flow in the control passage (90) when the pressure control mechanism (44, 46) raises the pressure in the crank chamber (15).

5. The compressor according to claim 1, **characterized in that** the selected chamber is the discharge chamber (38), wherein the control passage (100) allows gas to flow from the discharge chamber (38) to the crank chamber (15), wherein the controller controls the pressure adjusting valve (95) to restrict the flow of the gas in the control passage (100) when the pressure control mechanism (44, 46) raises the pressure in the crank chamber (15).

6. The compressor according to claim 1, **characterized in that**, when the pressure control mechanism (44, 46) increases the pressure in the crank chamber (15) to move the drive plate (31) to the minimum inclination position, the controller instructs the pressure adjusting valve (95) to regulate the control passage (90, 100) to limit the pressure in the crank chamber (15).

7. The compressor according to claim 6, **characterized in that**, when the compressor is stopped, the pressure control mechanism (44, 46) increases the pressure in the crank chamber (15) to move the drive plate (31) to the minimum inclination position.

8. The compressor according to claim 6, **characterized in that**, when the compressor is operating, the pressure control mechanism (44, 46) normally controls the pressure in the crank chamber (15) such that the drive plate (31) moves to an inclination position that corresponds to a desirable displacement, wherein, when a predetermined condition is satisfied, the pressure control mechanism (44, 46) increases the pressure in the crank chamber (15) to move the drive plate (31) to the minimum inclination position regardless of a desirable displacement.

9. The compressor according to claim 8, **characterized in that** an external drive source is connected to the drive shaft (16) to operate the compressor, wherein the predetermined condition is satisfied when there is a particular need to reduce the load applied to the external drive source.

10. The compressor according to claim 6, **characterized in that** the pressure control mechanism (44, 46) acts to move the drive plate (31) to the minimum inclination position and, simultaneously, the pressure adjusting valve (95) limits the pressure in the crank chamber (15).

11. The compressor according to claim 4, **characterized in that** the compressor includes a bleed passage (45, 90) that continuously connects the crank chamber (15) to the suction chamber (37) and permits gas to flow from the crank chamber (15) to the suction chamber (37).

12. The compressor according to claim 11, **characterized in that** the bleed passage serves as the control passage (90), wherein the pressure adjusting valve (95) limits gas flow in the control passage (90) when the pressure in the crank chamber (15) is appropriate.

Patentansprüche

1. Ein Kompressor mit veränderlicher Fördermenge, der Folgendes umfasst:

   ein Gehäuse einschließlich einer Zylinderbohrung (33), einer Kurbelkammer (15), einer Ansaugkammer (37) und einer Abgabekammer (38);

   einen Kolben (35), der in der Zylinderbohrung (33) untergebracht ist;

   eine Antriebswelle (16), die drehbar im Gehäuse gelagert ist;

   eine Antriebsplatte (31), die an den Kolben (35) gekoppelt ist, um eine Rotation der Antriebswelle (16) in eine Hin- und Herbewegung des Kolbens (35) umzuformen, wobei die Antriebsplatte (31) neigbar an der Antriebswelle (16) gelagert ist, wobei sich die Antriebsplatte (31) zwischen einer Maximalneigungsposition und einer Minimalneigungsposition entsprechend dem Druck in der Kurbelkammer (15) bewegt, wobei die Neigung der Antriebsplatte (31) den Hub des Kolbens (35) und die Fördermenge des Kompressors bestimmt;

   einen Drucksteuermechanismus (44, 46) zum Steuern des Drucks in der Kurbelkammer (15) zum Verändern der Neigung der Antriebsplatte (31), wobei der Drucksteuermechanismus (44, 46) ein Steuerventil (46) umfasst, das eine Magnetspule (66) hat; und

   eine Steuerdurchführung (90, 100) zum Verbinden der Kurbelkammer (15) mit einer gewählten Kammer im Kompressor, wobei der Kompressor
   gekennzeichnet ist durch:

   ein Druckeinstellventil (95), das in der Steuerdurchführung (90, 100) untergebracht ist, wobei das Druckeinstellventil (95) eine Magnetspule (95a) hat, wobei das Druckeinstellventil (95) eine Gasströmung in der Steuerdurchführung (90, 100) regelt; und

   eine Steuerung zum Steuern des Stroms, welcher der Magnetspule (95a) des Druckeinstellventils (95) zugeführt wird, um den Druck in der Kurbelkammer (15) zu begrenzen, um zu verhindern, dass der Druck in der Kurbelkammer (15) unerwünscht hoch wird.

2. Der Kompressor gemäß Anspruch 1,
   dadurch gekennzeichnet, dass
   der Kompressor ein Zwangselement (20) umfasst, das die Antriebswelle (16) in eine axiale Richtung zwingt, wodurch eine axiale Bewegung der Antriebswelle (16) begrenzt wird, wobei der Druck in der Kurbelkammer (15) die Antriebsplatte (31) dazu bringt, eine axiale Kraft auf die Antriebswelle (16) auszuüben, wenn die Antriebsplatte (31) in der Minimalneigungsposition positioniert ist, wobei die Steuerung das Druckeinstellventil (95) anweist, den Druck in der Kurbelkammer (15) so zu begrenzen, dass die axiale Kraft die Antriebswelle (16) nicht gegen die Kraft des Zwangselements bewegen kann.

3. Der Kompressor gemäß Anspruch 1,
   dadurch gekennzeichnet, dass
   der Drucksteuermechanismus (44, 46) eine Überdruckdurchführung zum Verbinden der Abgabekammer (38) mit der Kurbelkammer (15) umfasst, wobei das Steuerventil (46) in der Druckdurchführung vorgesehen ist, wobei das Steuerventil (46) eine Gasströmung von der Abgabekammer (38) zur Kurbelkammer (15) durch die Überdruckdurchführung steuert, wobei das Steuerventil im Wesentlichen die Überdruckdurchführung vollständig öffnet, um die Antriebsplatte (31) in die Minimalneigungsposition auf der Basis von Befehlen von der Steuerung zu steuern.

4. Der Kompressor gemäß Anspruch 1,
   dadurch gekennzeichnet, dass
   die gewählte Kammer die Ansaugkammer (37) ist, wobei die Steuerdurchführung (90) ein Strömen von Gas von der Kurbelkammer (15) in die Ansaugkammer (37) ermöglicht, wobei die Steuerung das Druckeinstellventil (95) öffnet, um eine Gasströmung in der Steuerdurchführung (90) zu erhöhen, wenn der Drucksteuermechanismus (44, 46) den Druck in der Kurbelkammer (15) anhebt.

5. Der Kompressor gemäß Anspruch 1,
   dadurch gekennzeichnet, dass

die gewählte Kammer die Abgabekammer (38) ist, wobei die Steuerdurchführung (100) ein Strömen von Gas von der Abgabekammer (38) in die Kurbelkammer (15) ermöglicht, wobei die Steuerung das Druckeinstellventil (95) so steuert, dass die Strömung des Gases in der Steuerdurchführung (100) begrenzt wird, wenn der Drucksteuermechanismus (44, 46) den Druck in der Kurbelkammer (15) anhebt.

6. Der Kompressor gemäß Anspruch 1, **dadurch gekennzeichnet, dass,** wenn der Drucksteuermechanismus (44, 46) den Druck in der Kurbelkammer (15) anhebt, um die Antriebsplatte (31) in die Minimalneigungsposition zu bewegen, die Steuerung das Druckeinstellventil (95) anweist, die Steuerdurchführung (90, 100) so zu regeln, dass der Druck in der Kurbelkammer (15) begrenzt wird.

7. Der Kompressor gemäß Anspruch 6, **dadurch gekennzeichnet, dass,** wenn der Kompressor angehalten ist, der Drucksteuermechanismus (44, 46) den Druck in der Kurbelkammer (15) anhebt, um die Antriebsplatte (31) in die Minimalneigungsposition zu bewegen.

8. Der Kompressor gemäß Anspruch 6, **dadurch gekennzeichnet, dass,** wenn der Kompressor in Betrieb ist, der Drucksteuermechanismus (44, 46) normalerweise den Druck in der Kurbelkammer (15) dergestalt steuert, dass die Antriebsplatte (31) sich in eine Neigungsposition bewegt, die einer gewünschten Fördermenge entspricht, wobei, wenn eine vorherbestimmte Bedingung erfüllt ist, der Drucksteuermechanismus (44, 46) den Druck in der Kurbelkammer (15) erhöht, um die Antriebsplatte (31) in die Minimalneigungsposition unabhängig von einer gewünschten Fördermenge zu bewegen.

9. Der Kompressor gemäß Anspruch 8, **dadurch gekennzeichnet, dass** eine externe Antriebsquelle an die Antriebswelle (16) angeschlossen ist, um den Kompressor zu betätigen, wobei die vorherbestimmte Bedingung erfüllt ist, wenn ein besonderer Bedarf besteht, die auf die externe Antriebsquelle wirkende Last zu reduzieren.

10. Der Kompressor gemäß Anspruch 6, **dadurch gekennzeichnet, dass** der Drucksteuermechanismus (44, 46) so agiert, dass die Antriebsplatte (31) in die Minimalneigungsposition gebracht wird und gleichzeitig das Druckeinstellventil (95) den Druck in der Kurbelkammer (15) begrenzt.

11. Der Kompressor gemäß Anspruch 4, **dadurch gekennzeichnet, dass** der Kompressor eine Ablassdurchführung (45, 90) umfasst, die kontinuierlich die Kurbelkammer (15) mit der Ansaugkammer (37) verbindet und ein Strömen von Gas von der Kurbelkammer (15) in die Ansaugkammer (37) ermöglicht.

12. Der Kompressor gemäß Anspruch 11, **dadurch gekennzeichnet, dass** die Ablassdurchführung als die Steuerdurchführung (90) dient, wobei das Druckeinstellventil (95) eine Gasströmung in der Steuerdurchführung (90) begrenzt, wenn der Druck in der Kurbelkammer (15) angemessen ist.

Revendications

1. Compresseur à déplacement variable comprenant :

un boîtier incluant un alésage de cylindre (33), un carter (15), une chambre d'aspiration (37), et une chambre de refoulement (38) ;

un piston (35) logé dans l'alésage de cylindre (33) ;

un arbre d'entraînement (16) supporté, pour tourner, par le boîtier ;

un plateau d'entraînement (31) couplé au piston (35) pour convertir la rotation de l'arbre d'entraînement (16) en mouvement alternatif du piston (35), le plateau d'entraînement (31) étant supporté pour basculer par l'arbre d'entraînement (16), dans lequel le plateau d'entraînement (31) se déplace entre une position d'inclinaison maximale et une position d'inclinaison minimale en fonction de la pression au niveau du carter (15), dans lequel l'inclinaison du plateau d'entraînement (31) détermine la course du piston (35) et le déplacement du compresseur ;

un mécanisme de commande de la pression (44, 46) pour commander la pression au niveau du carter (15) pour changer l'inclinaison du plateau d'entraînement (31), dans lequel le mécanisme de commande de la pression (44, 46) comprend une soupape de commande (46), qui a un solénoïde (66) ; et

un passage de commande (90, 100) pour raccorder le carter (15) à une chambre choisie dans le compresseur, le compresseur étant **caractérisé par** :

une soupape de réglage de la pression (95) située dans le passage de commande (90, 100), dans lequel la soupape de commande de pression (95) a un solénoïde (95a), dans lequel la soupape de commande de pression (95) régule l'écoulement du gaz dans le passage de commande (90, 100) ; et

un dispositif de commande pour commander le courant fourni au solénoïde (95a) de la soupape de commande de pression (95) pour limiter la pression au niveau du carter (15) pour empêcher que la pression au niveau du carter (15) n'atteigne une valeur élevée non souhaitable.

2. Compresseur selon la revendication 1, **caractérisé en ce que** le compresseur inclut un élément de poussée (20) qui pousse l'arbre d'entraînement (16) dans une direction axiale, qui restreint le mouvement axial de l'arbre d'entraînement (16), dans lequel la pression au niveau du carter (15) fait que le plateau d'entraînement (31) applique une force axiale à l'arbre d'entraînement (16) lorsque le plateau d'entraînement (31) a une position d'inclinaison minimale, dans lequel le dispositif de commande donne l'ordre à la soupape de commande de pression (95) de limiter la pression au niveau du carter (15) de telle sorte que la force axiale ne puisse pas déplacer l'arbre d'entraînement (16) en s'opposant à la force de l'élément de poussée.

3. Compresseur selon la revendication 1, **caractérisé en ce que** le mécanisme de commande de la pression (44, 46), inclut un passage de pressurisation pour raccorder la chambre de refoulement (38) au carter (15), dans lequel la soupape de commande (46) est située dans le passage de pressurisation, dans lequel la soupape de commande (46) règle un écoulement de gaz de la chambre de refoulement (38) au carter (15) par l'intermédiaire du passage de pressurisation, dans lequel la soupape de commande ouvre sensiblement complètement le passage de pressurisation pour déplacer le plateau d'entraînement (31) dans une position d'inclinaison minimale en fonction des instructions du dispositif de commande.

4. Compresseur selon la revendication 1, **caractérisé en ce que** la chambre choisie est la chambre d'aspiration (37), dans lequel le passage de commande (90) permet au gaz de s'écouler du carter (15) à la chambre d'aspiration (37), dans lequel le dispositif de commande ouvre la soupape de commande de pression (95) pour augmenter l'écoulement de gaz dans le passage de commande (90) lorsque le mécanisme de commande de pression (44, 46) élève la pression au niveau du carter (15).

5. Compresseur selon la revendication 1, **caractérisé en ce que** la chambre choisie est la chambre de refoulement (38), dans lequel le passage de commande (100) permet au gaz de s'écouler de la chambre de refoulement (38) au carter (15), dans lequel le dispositif de commande commande la soupape de commande de pression (95) pour restreindre l'écoulement du gaz dans le passage de commande (100) lorsque le mécanisme de commande de la pression (44, 46) élève la pression au niveau du carter (15).

6. Compresseur selon la revendication 1, **caractérisé en ce que**, lorsque le mécanisme de commande de la pression (44, 46) augmente la pression au niveau du carter (15) pour déplacer le plateau d'entraînement (31) en position d'inclinaison minimale, le dispositif de commande donne l'ordre à la soupape de commande de pression (95) de réguler le passage de commande (90, 100) pour limiter la pression au niveau du carter (15).

7. Compresseur selon la revendication 6, **caractérisé en ce que**, lorsque le compresseur est arrêté, le mécanisme de commande de pression (44, 46) augmente la pression au niveau du carter (15) pour déplacer le plateau d'entraînement (31) en position d'inclinaison minimale.

8. Compresseur selon la revendication 6, **caractérisé en ce que**, lorsque le compresseur fonctionne, le mécanisme de commande de la pression (44, 46) commande normalement la pression au niveau du carter (15) de telle sorte que le plateau d'entraînement (31) se déplace dans une position d'inclinaison qui correspond à une capacité souhaitable, dans lequel, lorsqu'une condition prédéterminée est satisfaite, le mécanisme de commande de la pression (44, 46) augmente la pression au niveau du carter (15) pour déplacer le plateau d'entraînement (31) en position d'inclinaison minimale quel que soit le déplacement souhaité.

9. Compresseur selon la revendication 8, **caractérisé en ce qu'**une source d'entraînement externe est raccordée à l'arbre d'entraînement (16) pour faire fonctionner le compresseur, dans lequel la condition prédéterminée est satisfaite lorsqu'il y a un besoin particulier de réduire la charge appliquée à la source d'entraînement externe.

10. Compresseur selon la revendication 6, **caractérisé en ce que** le mécanisme de commande de la pression (44, 46) agit pour déplacer le plateau d'entraînement (31) en position d'inclinaison minimale et, simultanément, la soupape de commande de pres-

sion (95) limite la pression au niveau du carter (15).

11. Compresseur selon la revendication 4, **caractérisé en ce que** le compresseur inclut un passage de purge (45, 90) qui raccorde de manière continue le carter (15) à la chambre d'aspiration (37) et permet au gaz de s'écouler du carter (15) à la chambre d'aspiration (37).

12. Compresseur selon la revendication 11, **caractérisé en ce que** le passage de purge fait office de passage de commande (90), dans lequel la soupape de commande de pression (95) limite l'écoulement du gaz dans le passage de commande (90) lorsque la pression au niveau du carter (15) est adéquate.

Fig.1

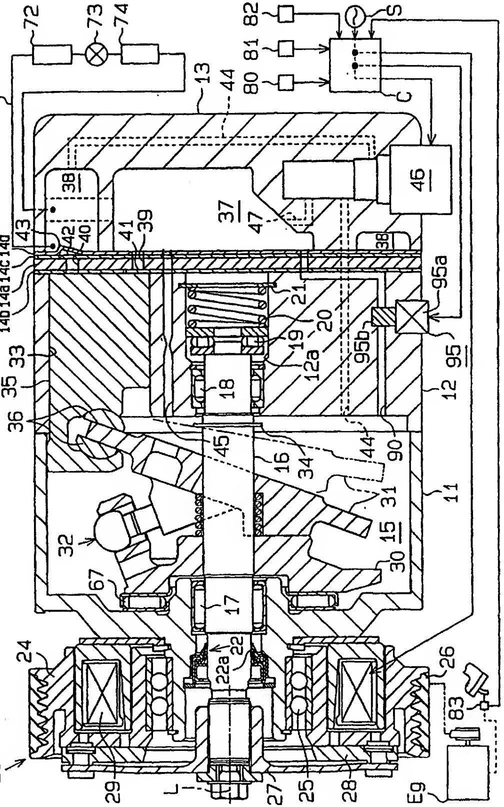

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Fig.2

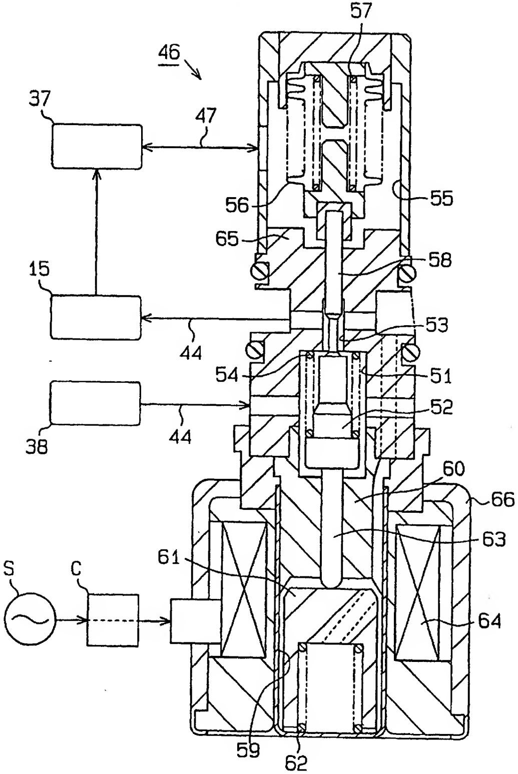

Fig.3

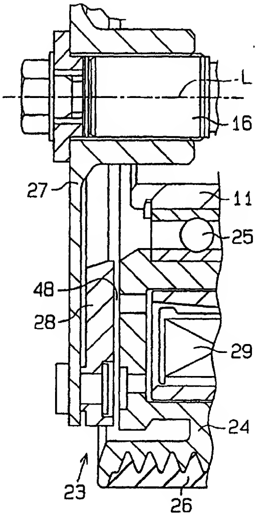

Fig.4

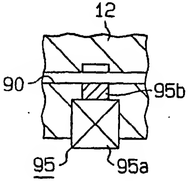

Fig.5

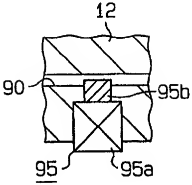

Fig.6

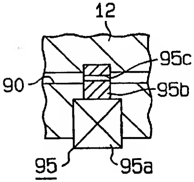

Fig.7

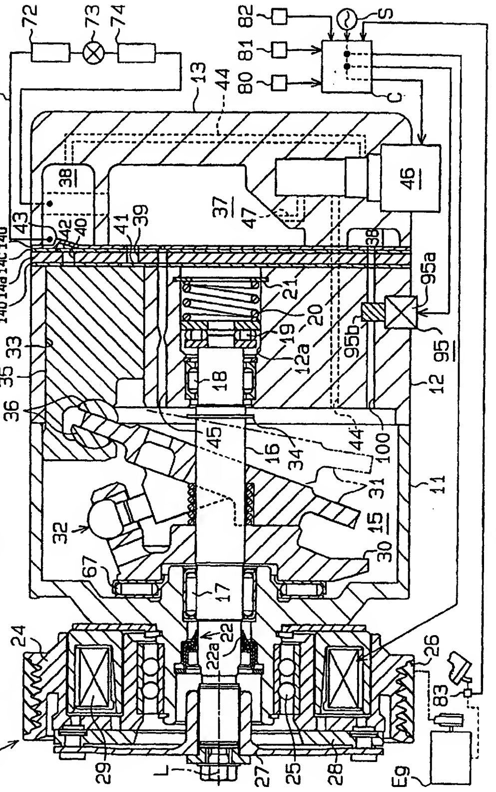

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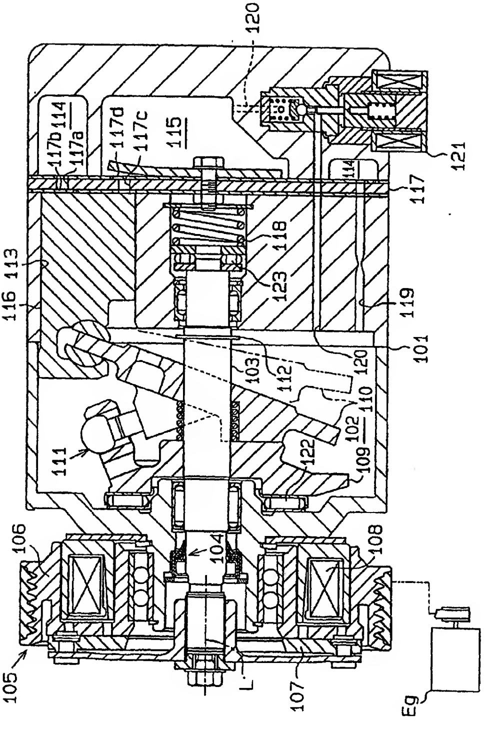
Fig.8



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